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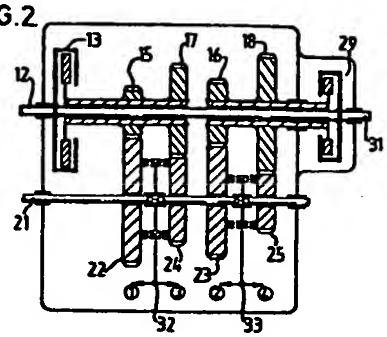
(54) Rotary power transmission.

(57) A rotary power transmission (11) has gear trains (15, 18), each being one of a set providing a series of increasing speed ratios, and two clutches (13,29) independently operable and arranged one at each side of the set of gear trains, the clutches providing alternative drive paths through the gear trains between a common input (12) and a common output (21), the gear trains of alternate speed ratios in the set are driven respectively through one and the other clutch.

FIG.2



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"Rotary Power Transmission"

This invention relates to a rotary power transmission of the dual clutch kind by which is meant a transmission having gear trains, each being one of a set providing a series of increasing speed ratios, and two clutches independently operable and providing alternative drive paths through the gear trains between a common input and a common output, the gear trains of alternate speed ratios in the set being driven respectively through one and the other clutch.

The invention is particularly though not exclusively applicable to drive transmissions for motor vehicles.

Transmissions of the dual clutch kind are derived from conventional single clutch manual-change transmissions with which they can share many component parts.

Examples of transmissions of the dual clutch kind are shown in British Patent Specification Nos. 145,827 (Bramley-Moore), 585,716 (Kegresse), 795,260 (David Brown) and 1,125,267 (Bosch). To the best of our knowledge, none of these prior

proposals has been commercially adopted.

For many years it has been customary for vehicle manufacturers to offer each vehicle model with the alternative of a manual-change or, where possible, a fully automatic transmission. As sales of vehicles fitted with automatic transmissions have risen it has become necessary for manufacturers to seek an automatic transmission for every new vehicle model.

Although most manufacturers use the same basic design of manual-change transmission, many different types of automatic transmissions are in use and most are complicated and carry a heavy cost disadvantage. Moreover each new type and model of automatic transmission requires an expensive test programme to ensure satisfactory strength and functioning, requires the manufacture and stocking of many new parts and may require the training of personnel to service the transmission.

Conventional automatic transmissions frequently take up more space than their manual-change transmission counterparts. The trend towards front engine-front wheel drive vehicles has meant that, in the smaller vehicle ranges, no

automatic transmission option is offered simply because sufficient space to install the transmission is not available.

Transmissions of the dual clutch kind have not come into use despite having the advantage of simplicity and of component part interchangability with the equivalent manual-change transmissions.

All of the previous proposals for transmissions of the dual clutch kind have the clutches arranged at the same side of the set of gear trains, a single driving member being provided for both clutches in most cases.

According to the present invention in a rotarypower transmission of the dual clutch kind, one clutch is arranged at one side of the set of gear trains and the other clutch is arranged at the opposite side of the set.

A major advantage of this arrangement is that a manual-change transmission, from which a transmission of the dual clutch kind is adapted, can remain substantially unchanged and the spatial problems of installation are considerably redu-ed as compared with an arrangement of a dual clutch at one side of the set of gear trains. The

external dimensions of a manual-change transmission can remain virtually unchanged except for the provision of an additional clutch which may be located co-axial with the transmission input shaft or with a transmission layshaft.

A subsidiary advantage of arranging the clutches on either side of the set of gear trains is that the design of control connections to each clutch is eased simply because the clutches are not adjacent.

Variations are possible in the selection of the type of clutch provided at one side or the other of the set of gear trains, it being possible to suit particular requirements by selecting, for example, a dry plate friction clutch for drive take-up from rest.

However, the additional clutch is preferably used only for speed ratio changes in the transmission and not for drive take-up from rest. As a result the required capacity of the additional clutch may be much reduced and the consequent benefit of small size together with the aforementioned capability of alternative location within the transmission ensures that space for the clutch

can be found in existing installations.

One clutch maybe a dry plate friction clutch and is advantageously the same as that specified for an equivalent manual-change transmission. Preferably this clutch is used for drive take-up from rest only for the gear train of the lowest forward speed ratio.

The necessary ancilliary controls for the transmission may be situated in the transmission casing or in any convenient space in the installation site.

The additional clutch maybe a fluid-pressure engaged wet friction clutch. Such a clutch can operate in the transmission fluid and so does not require to be sealed from the main transmission casing.

Adoption of a transmission of the dual clutch kind according to the invention will enable a vehicle manufacturer to take advantage of a number of subsidiary advantages. Interchangability of parts with equivalent manual-change transmission will enable the manufacturer to achieve economies of scale, these economies being substantial for relatively expensive components such as gear wheels.

The additional cost of testing and proving a transmission of the dual clutch kind will be low as compared with the costs for a new design of conventional automatic transmission.

The manufacturer and vehicle dealer can reduce their spares inventory; servicing of the transmission should be within the capabilities of a skilled mechanic.

Finally, since it is anticipated that the cost of manufacture and service support of a transmission of the dual clutch kind provided by the invention will be lower than that of a conventional automatic transmission, the cost to the buyer of specifying an automatic transmission option should be much reduced.

Other features of the invention are included in the following description of two preferred embodiments shown, by way of example only, on the accompanying drawings in which:-

Fig. 1 illustrates a conventional two shaft manual-change transmission having gear trains in constant mesh;

Fig. 2 illustrates one embodiment of the invention as applied to the transmission of Fig. 1; and

Fig. 3 illustrates an alternative embodiment of the invention as applied to the transmission of Fig. 1.

With reference to Fig. 1 there is shown a conventional four speed gear transmission 11 for interposition in a vehicle drive line between the engine and the driven wheels. The arrangement is particularly suitable for vehicleshaving transversely mounted engines with front wheel drive, the transmission output being on the same side of the set of gear trains as the input.

An input shaft 12 is connectable through a disengagable clutch 13 to a transmission main shaft 14. The main shaft 14 carries fast thereon gear wheels for rotation therewith,respectively for first speed ratio 15, second ratio 16, third ratio 17 and fourth ratio 18.

A countershaft 21 has journalled thereon gear wheels 22-25 for individual rotation, each respectively in mesh with one of the wheels 15-18 of the main shaft 14.

Synchroniser assemblies 26, 27 are movable axially by a transmission change speed lever to connect the countershaft 21 to any one of the gear wheels 22-25.

The countershaft 21 is the output shaft of the transmission and is connected to the differential gear of the vehicle drive.

Reverse speed ratio may be obtained by any known means, for example, the use of a separate gear train having a lay gear to reverse rotation of the drive. Such an arrangement may be found in many manual-change transmissions.

The operation of such a gear transmission is well known but for clarity a ratio change from second speed to third speed will be described.

Synchroniser assembly 26 is in the rightwards position, as viewed, to connect the main shaft 14 to the countershaft 21 through second speed gear train 16, 23. The clutch 13 is engaged and driving torque is transmitted from the input shaft 12 to the countershaft 21. The first, third and fourth speed gear trains are driven idly at engine speed by the main shaft 14.

To change speed ratio the clutch 13 is disengaged, so breaking the driving torque transmitted through the transmission. Synchroniser assembly 26 is moved to disconnect second speed gear wheel 23 from the countershaft and third speed gear wheel 24 is connected to the countershaft 21 by moving synchroniser assembly 27 leftwards, as viewed.

The ratio change is completed by re-engaging the clutch 13 to re-establish driving connection between the engine and the vehicle driven wheels. First, second and fourth speed gear trains are now driven idly by the main shaft.

Although the manual-change transmission has been described with the relatively rotatable gear wheels journalled on the countershaft 21, some or all of the gear trains could have the relatively rotatable gear wheels, under the control of the synchroniser assemblies, supported by the main shaft with the respective meshing gear wheels being fast for rotation with the countershaft. This arrangement would result in some or all of the non-driving gear trains being driven idly at road speed.

The transmission of Fig. 2 and 3 has the position of second and third speed gear trains inter-changed so as to place the "odd" ratios, first and third, and the "even" ratios, second and fourth, in adjacent pairs.

The transmission main shaft is tubular and divided into two parts so that the "even" gear trains and "odd" gear trains can be separately driven from the input shaft 12.

Referring to Fig. 2, the "odd" gear trains are driven through the clutch 13, and the "even" gear trains through an additional clutch 29 which is itself driven from the input shaft 12 by a shaft 31 co-axial with and extending through both parts of the tubular main shaft.

It is intended that automatic speed ratio selection would be effected using known automatic transmission control technology as described for example in 'Torque Converters or Transmissions' by P.M. Heldt and published by Chilton. The transmission would be provided with a selector lever having the usual DRIVE, NEUTRAL, REVERSE and PARK positions.

Operation of the transmission is as follows:-

The vehicle engine is running, the selector lever is in NEUTRAL and both clutches 13 and 29 are disengaged.

The vehicle driver engages DRIVE and the transmission automatic control selects first speed by moving synchroniser assembly 32 to connect gear wheel 22 to the countershaft 21. As the driver accelerates the engine, the clutch 13 is automatically engaged and driving torque transmitted through gear wheels 15, 22, to the vehicle drive.

The transmission automatic control can now select second speed by moving synchroniser assembly 33 to connect gear wheel 23 to the countershaft 21. Consequently the second ratio drive train of gear wheels 23, 16 and the driven plate of clutch 29 are driven idly by countershaft 21, the drive member of clutch 29 being driven constantly at engine speed by the shaft 31.

The transmission automatic control initiates a ratio change from first speed to second speed in response to the values of the usual

control parameters such as throttle opening and road speed.

The clutch 13 is disengaged as the clutch 29 is engaged and driving torque is progressively transferred from the first ratio gear train to the second ratio gear train so providing a "power on" ratio change. When the change is completed the first ratio drive train is driven idly by the countershaft 21.

Dependent on the value of the automatic control parameters the synchroniser assembly 32 will remain with first speed ratio selected or be moved to select third speed ratio by connecting gear wheel 24 to the countershaft 21. Alternatively synchroniser assembly 32 may shift to the neutral position until the automatic control indicates an imminent ratio change.

Thus the transmission automatic control is arranged to pre-select the next required speed ratio in sequence, the ratio change being automatically made at the appropriate moment by changing drive from one clutch to the other.

It is not intended that the additional clutch 29 should be used for drive take up from rest. As a result the required capacity is much reduced and the clutch can be of small size when compared with the main clutch 13.

With reference to Fig. 3 there is shown a modification of the dual clutch transmission where the additional clutch 29 is co-axial with the countershaft 21.

The operation of the transmission is as previously described for the embodiment of Fig. 2. Speed ratios are pre-selected by the automatic control in accordance with the usual control parameters and, at the appropriate moment, the automatic control effects a ratio change by disengaging the drive through one clutch as the drive through the other clutch is engaged.

Although the invention has been described with reference to a constant mesh gear arrangement, it is equally adaptable to other gear arrangements such as, for example, those utilising sliding gears or sliding dogs.

Claims

1. A rotary power transmission having gear trains (15,18), each being one of a set providing a series of increasing speed ratios, and two clutches (13,29) independently operable and providing alternative drive paths through the gear trains between a common input (12) and a common output (21), the gear trains of alternative speed ratios in the set being driven respectively through one and the other clutch, characterised thereby that one clutch (13) is arranged at one side of the set of gear trains and the other clutch (29) is arranged at the opposite side of the set.

2. A transmission according to Claim 1, characterised thereby that one clutch (13) is a spring engaged, dry plate friction clutch engagable to provide a drive path through the gear train of at least one odd speed ratio (15,17).

3. A transmission according to Claim 2, characterised thereby that said one clutch(13) is engagable for drive take-up from rest only for the gear train (22) of a lowest forward speed ratio.

4. A transmission according to Claim 2, characterised thereby that said one clutch (13) is engagable to provide a drive path through a gear train of reverse speed ratio.

5. A transmission according to Claim 2, characterised thereby that the other clutch (29) is a fluid-pressure engaged, wet friction clutch engagable to provide a drive path through the gear train of at least one even speed ratio (16,18).

6. A transmission according to any preceding Claim , characterised thereby that both clutches (13,29) precede at least one respective gear train of a speed ratio in the drive path between said input (12) and said output (21).

7. A transmission according to any preceding Claim, characterised thereby that the clutches (13,29) are co-axial.

8. A transmission according to Claim 7, characterised thereby that one clutch (29) has a drive shaft (31) extending from the input (12) through the other clutch (13).

FIG.1.

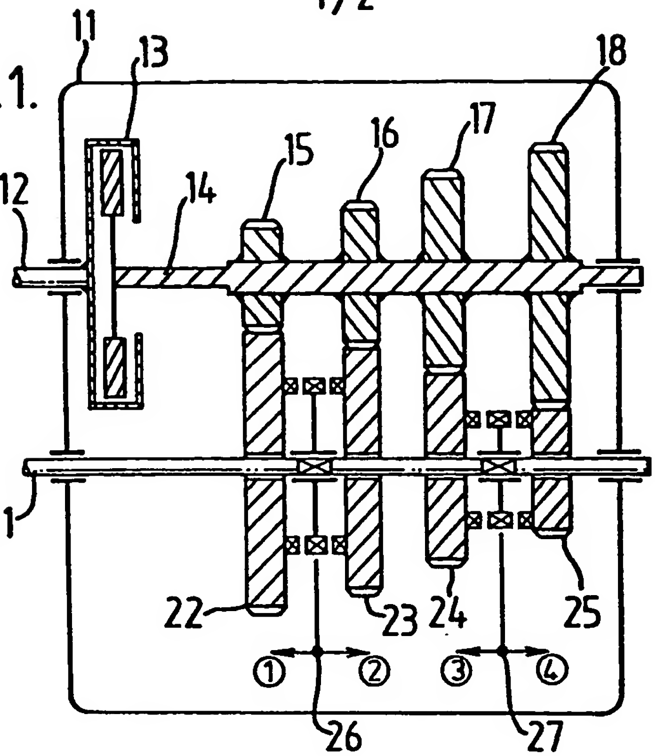
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FIG.2.

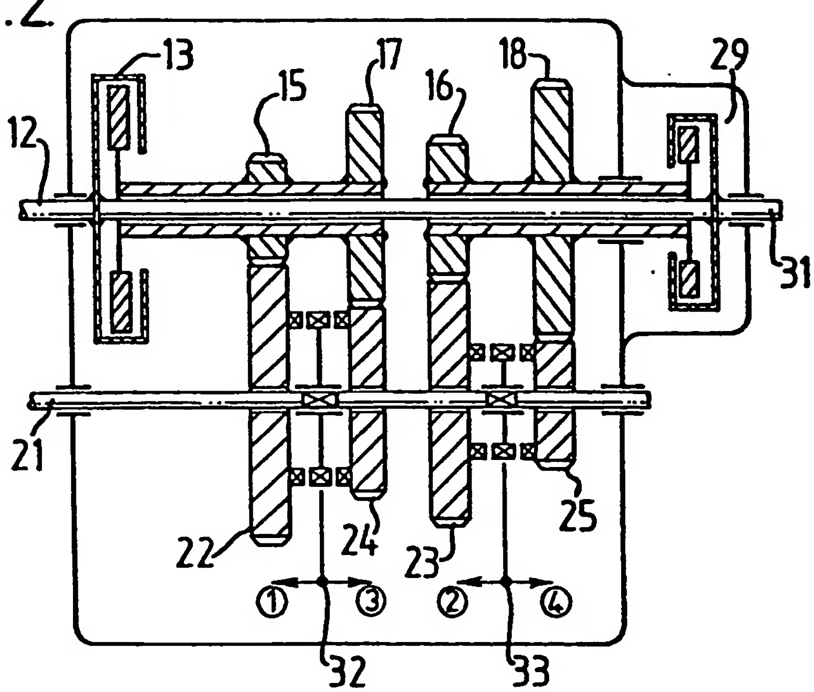
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FIG. 3.